specification and claims and from the accompanying drawings which illustrate an embodiment of the invention.

FIGS. 1–7 are schematic representations of the relative motion which takes place between the blades of an eight blade rotor, the vanes of a six vane stator and the two lobed spinning mode created thereby, which mode is rotating in a counterclockwise direction.

FIGS. 8–17 are schematic representations of the relative motion which takes place between the blades of an eight blade rotor, the vanes of a nine vane stator and the single lobed spinning mode created thereby, which single mode is rotating in a clockwise direction.

FIG. 18 is an external side view of a typical turbo-jet engine of the type used in a modern aircraft and which will utilize our invention.

FIG. 19 is a side view of a shrouded marine propeller, partially broken away to illustrate our invention.

FIG. 20 is an enlarged, cross-sectional view showing of the forward end of an axial flow compressor illustrating a preferred embodiment of our invention.

FIG. 21 is a graph illustrating the noise reduction which can be expected by axially spacing adjacent stators and rotors various distances. The graph scale chosen is 1 inch=1 chord length.

FIGS. 22, 23, 24 and 25 illustrate the decay rates of spinning modes having various lobe numbers m and which rotate at various wall speeds or Mach numbers. These figures illustrate the effects of decay when the hub-tip ratio is 0, 0.25, 0.50 and 0.75, respectively.

FIGS. 26, 26*a*, 26*b* and 26*c* are graphic representations of various solutions of Formula 1 used to illustrate how blade and vane combinations are selected to avoid spinning mode propagation and insure spinning mode decay within the compressor duct for various Mach numbers.

FIG. 27 is a typical far field spectrum of an axial flow compressor designed to decay the fundamental (B) and the first harmonic (2B), with the higher harmonic (3B etc.) propagating.

FIG. 28 is a schematic representation of cantilevered inlet guide vanes 59 projecting radially inwardly from the compressor outer case and extending throughout the blade tip region only so as to satisfy both the requirement of high vane number and object ingestion.

FIG. 29 is a compressor having retractable inlet guide vanes.

FIG. 30 is a developed view projected, both axially and circumferentially, illustrating a spinning mode and indicating the ridges of high and low pressure.

FIG. 31 is a graphic representation of a developed view of two spinning modes created by different sources in an axial flow compressor, which modes are illustrated precisely as generated.

FIG. 32 is similar to FIG. 31 but shows the modes in a corrected condition 180° out-of-phase and hence cancelling.

FIG. 33 is a developed view of a two-lobe mode and a three-lobe mode as generated in an axial flow compressor.

FIG. 34 corresponds to FIG. 33 but shows the two-lobe mode cancelled.

FIG. 35 corresponds to FIGS. 33 and 34 and shows both the two-lobe mode and the three-lobe mode cancelled.

FIG. 36 is an illustration of the inlet end of a compressor and the parameters used in considering the directivity pattern created by the spinning modes propagating therefrom.

FIGS. 37–40 illustrate the directivity pattern for a single lobed spinning mode as engine speed is increased.

FIGS. 41 and 42 illustrate a comparison of test data and theoretical data.

FIG. 43 is a graphic representation of the ground exposure time of directivity patterns established by spinning modes having various numbers of lobes m.

FIG. 44 is the Fletcher-Munson curve. This shows the

variation of ear sensitivity to noise as a function of frequency.

FIG. 45 is a graphic illustration of the extra air attenuation for noise generated at various frequencies.

FIG. 46 is a partial enlarged showing of a portion of the compressor shown in FIG. 20 and illustrating adjacent stators and rotors and their connection to permit axial, circumferential and helical indexing.

FIG. 47 is a developed view of the face spline connections of FIG. 46.

FIG. 48 is a showing of the relation between the number of vanes and shaft speed for an illustrative engine to meet the requirements of the indicated noise criteria.

FIGS. 49 and 50 show families of curves illustrating blade-vane combinations suitable for various operating speeds so as to satisfy the designated acoustic requirements.

In view of the complexity of the theory of the derivation and behavior of discrete frequency compressor noises, it will be necessary to use mathematical formulae and symbols throughout this description. According, Appendix A is provided at the end of the description to provide a table of definitions of the various symbols to be used herein.

THE NATURE OF COMPRESSOR NOISE

Since our initial study of the noise problem was directed toward solving the noises in turbojet aircraft engines, our description will be directed primarily toward the noise problems as they apply to a turbojet aircraft engine, but it should be borne in mind that there are other equally important applications which may be found in all types of rotating machinery of the vane-blade type, such as the marine propeller application referred to herein.

The nature of compressor noise is described in substantial particularity in an article by J. M. Tyler and T. G. Sofrin entitled "Axial Flow Compressor Noise Studies" appearing in SAE Transactions, Reprint, 1962, pages 309 through 332, which is hereby incorporated by reference and to which particular reference may be had.

As more fully developed in the aforementioned SAE article, there are two sources of the discrete frequency compressor noise, consisting of the fundamental blade-passage frequency and the harmonics of the first one or two stages. These sources are (1) the rotation of the periodic pressure field of the blades operating in uniform flow, and (2) the periodic cutting by the rotor of flow irregularities produced by stator vanes or the periodic impingement of rotating blade wakes on stator vanes.

Both sources produce what have been called spinning modes which are periodic pressure patterns that spin about the shaft axis. However, patterns produced by the two sources differ in an important way. In the case of the first source; a rotor operating in uniform flow, the pattern associated with the order nB consists of nB lobes, or cycles of simple harmonic variation, rotating at exactly rotor speed N, where B is the number of blades and n is the harmonic being considered, with $n=1$ for the fundamental, $n=2$ for the first harmonic and so forth. Thus, the fundamental blade-pasage frequency, BN, is generated by a B-lobe pattern turning at rotor speed, the first harmonic, 2BN, is associated with a pattern having 2B lobes turning at rotor speed, and so forth. As fully explained in the aforementioned SAE article, rotor-created spinning modes can be caused to decay rapidly by operating the rotor below cut-off speed according to the formula:

$$\frac{k'^{(\sigma)}_{m\mu}}{n}$$

where $m=nB$, n=harmonic index, B equals number of blades, and

$$k'^{(\sigma)}_{m\mu}$$

is a characteristic number.